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Compressor Tolerance to Liquid Refrigerant

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COMPRESSOR TOLERANCE TO LIQUID REFRIGERANT

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INTRODUCTION

The most singular item which most frequently can destroy a refrigerant compressor is liquid refrigerant. Many other items such as high voltage, a capacitor failure, extreme operating conditions and system contamination often result in compressor failures. Analysis of these failures indicates that they generally require long periods of time to develop and often may be detected and corrected before the failure actually occurs. This is not true if the compressor is allowed to digest liquid refrigerant. As with poison or drugs in a human, the first time may be the last.

The study of the tolerance of a compressor to liquid refrigerant as presented here is confined to hermetic compressors. This is the type of machine where both the compressor and the driving motor are enclosed in a welded steel shell. Most of the test work has been performed on 3-ton and smaller machines, although some of the evaluation has been with compressors up through 20 tons of capacity. This material is meant to only emphasize the importance of properly evaluating refrigerant floodback and migration conditions during the initial design phases of a new machine. Naturally, these evaluations must be married with the other major design requirements to yield a totally acceptable machine. It is not the intent of the author to solve all liquid refrigerant return problems. Presented are test equipment outlines, test procedures, and results experienced during evaluations of floodback performance of current and new design compressors. Each and every compressor design in the industry requires its own particular configuration to satisfy the tolerance levels recommended.

The ASHRAE Guide, manufacturers system specifications, and various articles in trade journals have fairly well covered system considerations of preventing liquid refrigerant from entering the compressor. Some of these are: the application of crankcase heaters, the use of suction line accumulators, the requiring of pump down cycles and/or the imposing of limitations upon the maximum system charge. Although all the above controls are effective, the compressor engineer must recognize that none are foolproof. In today's world, we find that the probability of liquid refrigerant being present in the compressor at some time during the life of the system is high. For example, the lead of a crankcase heater may vibrate loose or the heater itself may fail. With the loss of crankcase heat, the system charge is very likely to migrate into the compressor shell during a system off period. Upon start-up, the refrigerant oil mixture will be drawn directly into the compressor and possibly cause a failure.

From a compressor design viewpoint, and from the effects on manufacturers' warranty claim payments, it is not practical to allow this condition to result in the termination of the compressors useful life.

CAUSES OF A FLOODED START

With this introduction the question arises, "What else may cause the refrigerant to migrate to the compressor shell". The primary cause of extreme migration is a loss of the evaporator load. This loss can be the result of the failure of the evaporator fan motor or a fan belt breakage, an evaporator coil freeze-up, an expansion valve failure, and/or plugged return air filters. Any of these items will result in the loss of the evaporator load and lead to the direct return of the liquid refrigerant from the condenser to the compressor. This in itself, if not properly considered in the compressor design, may cause a failure.

If the raw liquid return does not fail the compressor, a still more severe condition may exist upon the next compressor start. This is the immediate ingestion of the entire system charge into the compressor after it has accumulated in the shell. Prior to this start, the previously mentioned returning raw refrigerant has cooled the compressor shell and internal parts to possibly -20^{o}F. During the off cycle the compressor is the coldest component of the system and therefore represents the lowest saturated pressure level. It then stands to reason that any refrigerant in the other system components will be forced to travel (migrate) to compressor. When a start-up occurs, this liquid refrigerant is drawn directly into the cylinders which results in extremely high discharge pressure. As a side light, the meager heat of a crankcase heater will only tend to

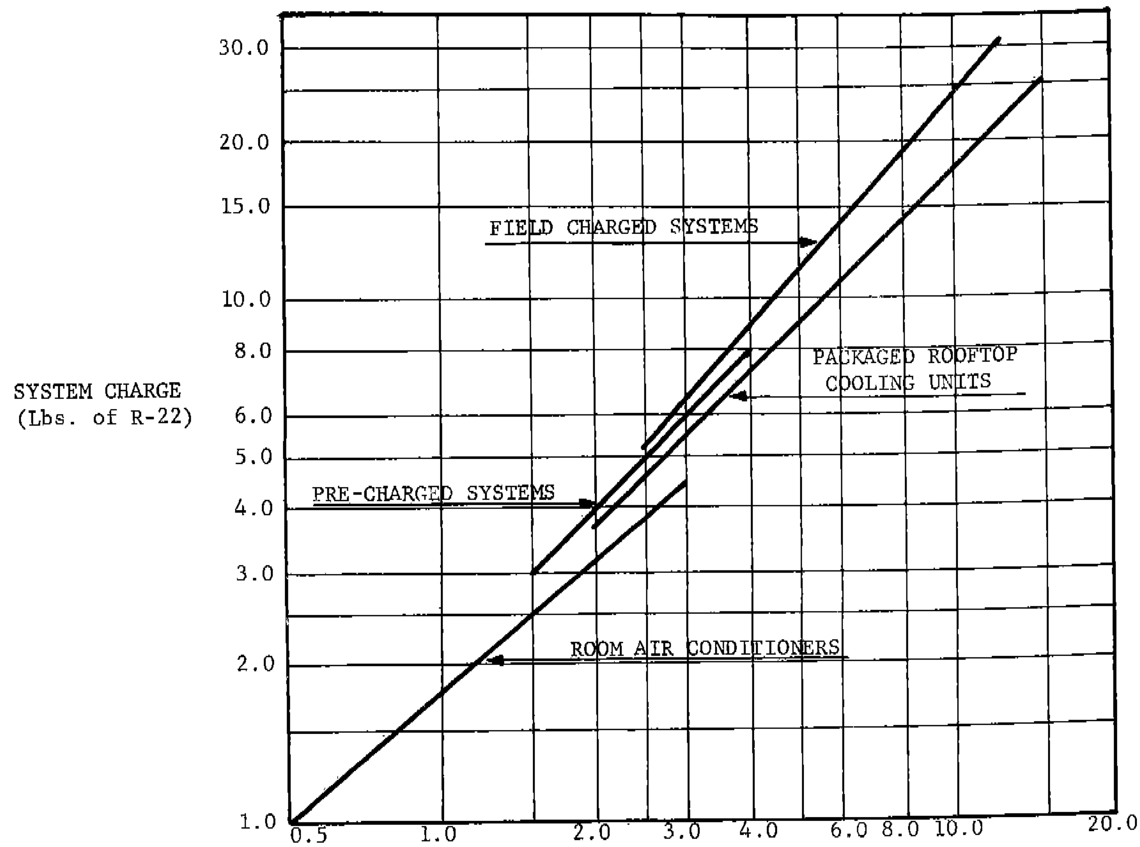

FIGURE I
Nominal System Charge Requirements

slowly raise the saturated pressure level in the compressor. Crankcase heat will not drive refrigerant from the compressor until the shell temperature is greater than that of another system component and this may require several hours to take place.

Other system operating malfunctions beside the loss of evaporator control can cause the same condition - the entire system charge being present in the compressor shell. Some of these are: the condenser coil airflow blockage on a capillary tube feed system, improper field charging procedures, a bypass control failure, and/or change-over season operation.

EFFECTS OF A FLOODED START

It has now been established that liquid can be present in the compressor shell upon the next start-up. But what happens when this condition exists? Why will this, if not properly anticipated in the compressor design, result in such a catastrophe and immediate failure? Grossly, what takes place is that a compressible fluid machine has been applied on an incompressible medium. Due to the compressible gas relationship sizing of discharge passages at the same volume flow rate, extremely high pressure levels will be developed (2500 psi). These pressures will result in the physical destruction of valve plates, gasket, mufflers, connecting rods and loss of assembly alignment. Any of these items will render the machine inoperative. Some side effects of large quantities of liquid refrigerant in the shell are oil stratification, de-greasing, and high refrigerant/oil ratios. All of which may result in insufficient bearing lubrication and the scoring or seizure of the bearings.

With this type of future for his compressor, the engineer is forced to give major consideration to this problem in his initial design. The first item to determine is how much liquid refrigerant will be present. A conservative design criteria is to assume that 100% of the system charge may be present in the compressor shell upon start-up. System charge is primarily a function of capacity and type of system in which the compressor is to be applied. Shown on Figure 1 are representative system charges for various types of systems from one-half to twenty tons of nominal capacity. As previously stated, the designer must not allow these charge levels to result in a compressor failure Therefore, the chart in Figure 1 may be considered as a minimum allowable compressor liquid refrigerant tolerance chart for a practical compressor design.

TEST EQUIPMENT FOR EVALUATING

Utilizing this background as design requirements, facilities have been built to run accelerated tests of both purchased and internally manufactured compressors as shown in Figure 2. A second variation of these facilities has been also used which does not utilize an evaporator and throttles the liquid refrigerant directly back to the compressor.

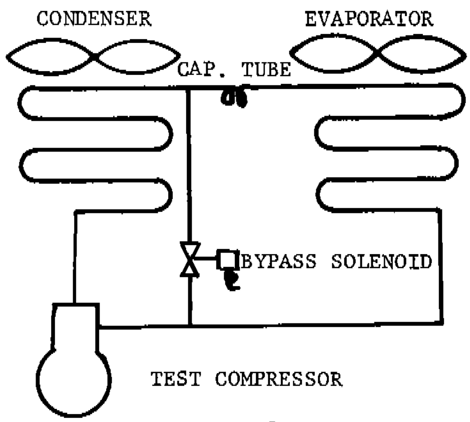

FIGURE 2
Test Facility Refrigerant Circuit

The compressor to be tested is installed in a representative air conditioning system using an air cooled condenser. The evaporator (if used) is located higher than the compressor to promote the natural drainage of liquid to the compressor. A solenoid valve is connected as a bypass between the condenser outlet and the compressor suction line for immediate system balancing upon shut down. The system is charged to the proper level as shown in Figure 1 and operated without a crankcase heater. Timers and relays are used to sequence the operation of the components as follows:

Step 1: 15 seconds no operation.
Step 2: 15 seconds only the compressor operates.
Step 3: 9 minutes, 30 seconds the compressor and condenser fan only operate.
Step 4: 10 minutes compressor off, bypass valve open, condenser and evaporator fans on.
Step 5: Return to Step 1.
Step 6: Optional long overnite off cycle no components operating, return to Step 1.

Additional tests follow by increasing the system charge in one pound increments until the limit is determined at which severe damage is imminent. At this point, the system is recharged to the previous test quantity and run for 500 hours system of running time. During this period no damage should result to the compressor. This produces a minimum of 1000 starts with liquid digestion and borderline lubrication.

Failures experienced during this phase of new compressor development has initiated interest with respect to the actual pressure levels developed during severe liquid digestion (slugging). To observe these rapid pressure pulses, pressure transducers were installed in the cylinder cover of the compressor. The output of the pressure transducer was displayed on an osciloscope and photographed for record. The osciloscope was syncronized with the compressor with a magnetic pickup on the crankshaft, and a second trace was displayed to indicate shaft position. The hardware of this instrumentation is shown in Figure 3.

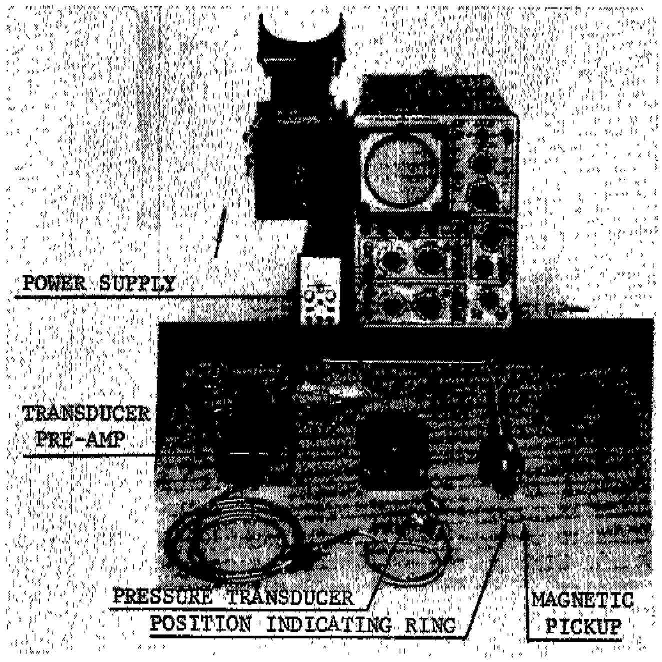

FIGURE 3
Pressure Transducer Instrumentation

During test runs with this instrumentation, the first audible indications of slugging have been observed when the peak pressure levels are approximately 1600 psi (Figure 4).

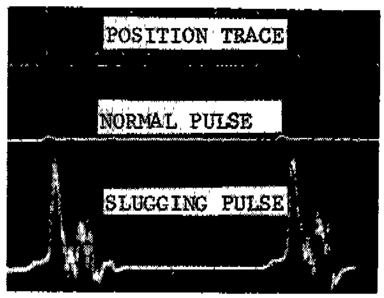

FIGURE 4
Slugging Pressure Pulse (1605 psi);
Cylinder Cover Pressure Wave

It was hoped that the pressure transducer display would indicate slugging before audible signs were present. This has not been observed. The maximum pressure peaks observed during a severe slugging period have been in the range of 2500 psi

(Figure 5). Hydrostatic tests of high side parts have verified these pressure levels when comparing pressure levels required for the component deformation experienced.

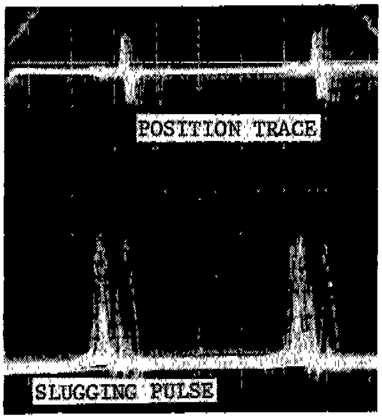

FIGURE 5
Slugging Pressure Pulse (2205 psi max.), Cylinder Cover Pressure Wave

The peak pressure levels do not occur during the first few revolutions of the test machines. Peak pressures are developed only after approximately 30 to 60 shaft revolutions. This is when sufficient inertia has been developed in the rotating parts to carry the extremely high loads imposed by the high piston pressures. Also, sufficient quantity of fluid has been pumped to result in the choking of flow in the discharge passages and cause the high back pressures on the piston. During the extreme slugging periods, there is reduction of shaft speed as the rotating inertia is lost and the motor does not have sufficient torque to maintain the normal speed. This shaft speed reduction is noted in Figure 5 by the slipping of the position indicating trace. The period of the pressure pulse is only of a very short duration (approximately .003 seconds) but often may be repeated 10 to 15 times per start.

CORRELATION

Some of the catastrophic effects of slugging may be seen in the photographs of failed parts. To ballon the steel discharge muffler as shown in Figure 6, would require 2200 psi during a hydrostatic pressure test. This muffler was so deformed in one slugging period during developmental slug testing on the test facility previously described.

The discharge valve plate (Figure 7) was dished and fractured in the same facility with a test refrigerant charge of 5 lbs. Relating the capacity of the tested compressor to Figure 1 indicates the compressor should be able to tolerate 6 to 7 lbs. of liquid refrigerant.

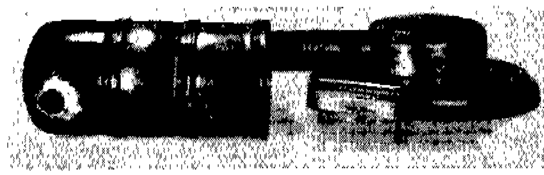

FIGURE 6
Slug Test Deformed Steel Discharge Muffler

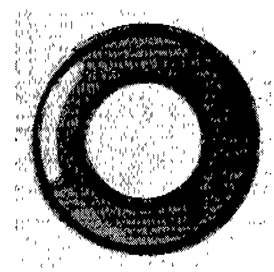

FIGURE 7
Slug Test Fractured Discharge Valve Plate

The stretched inner discharge valve seat rivet, (Figure 8) was also failed during developmental slug testing. A 1520 lb. force is required to yield this rivet to the degree shown. When considering the relationship between cylinder cover pressure and cylinder pressure (the difference across the discharge valve) a 2100 psi cylinder cover pressure would be required to develop the 1520 lbs. of force required for the failure.

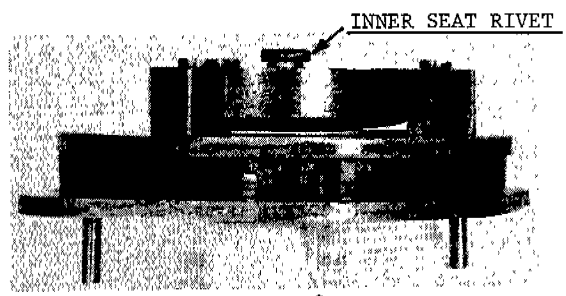

FIGURE 8
Slug Test Failed Inner Seat Rivet

The fractured cast iron discharge muffler (Figure 9) was removed from a customers failed compressor. From Figure 1 this compressor should be designed to tolerate 4.5 lbs. of liquid refrigerant. The actual factory system charge for the particular unit is 3.6 lbs. Naturally, there is some question as to whether the system was properly charged at the factory or improperly recharged in the field. The example is only included to illustrate the severe nature of a slugging condition.

The final example is one of the marginal lubrication experienced during a flooding condition. These bearings (Figure 10) are the main crankshaft bearings of a developmental compressor. They were failed on the floodback facility during a slugging start. During a slugging start not only are the bearings exposed to a poor lubricating medium but also the severe piston back pressure load is present. This combination caused the failure shown.

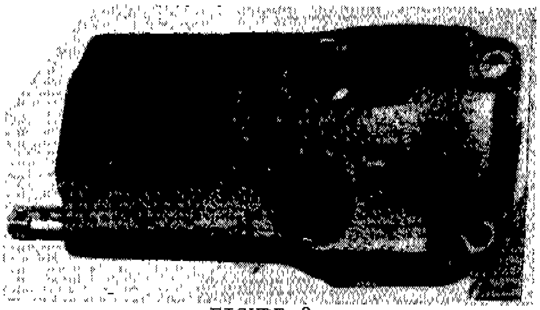

FIGURE 9
Field Fractured Discharge Muffler

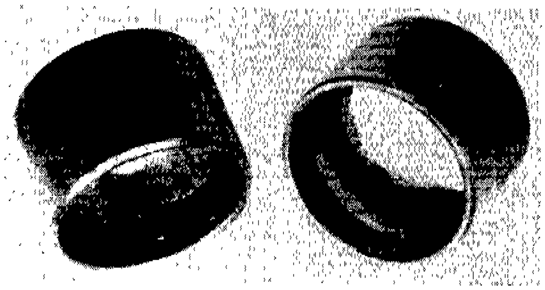

FIGURE 10
Failed Main Crankshaft Bearing
As The Result Of
Liquid Refrigerant Washing and Slugging Loads

SUMMARY

The basic emphasis of this paper has been to illustrate the severe destructive nature of liquid refrigerant to a compressor. Therefore, it is of prime importance to the compressor engineer to equate these effects to his particular configuration. It is recommended that the entire system charge should be included in these evaluations as presented in Figure 1. At the same time, the system engineer must design for minimum system charge when selecting <u>system</u> components. Often the possible detrimental effects on the life of the compressor are ignored when selecting an oversized suction line or liquid line.

From the data presented, it is recommended that all internal high side compressor parts should be designed to withstand pressure differentials in excess of 2000 psi. Also highly restrictive discharge side passages which are immediately downstream of the piston are to be avoided so that the instantaneous slugging pressure pulses previously illustrated can be readily disipated.